

Short Course on Experimental Dynamic Substructuring

Module #4: Exercises



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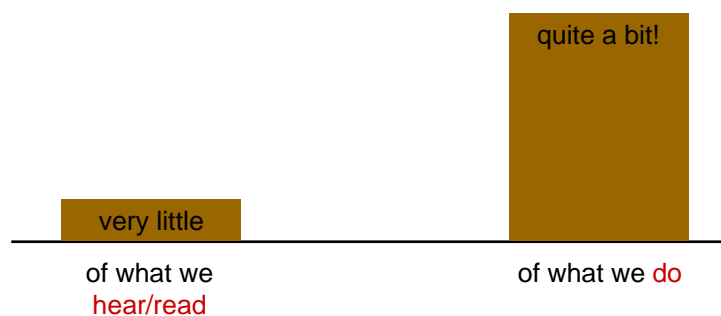
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Short Course Notes For:

February 1, 2014, IMAC, Orlando, Florida

The Science of Learning

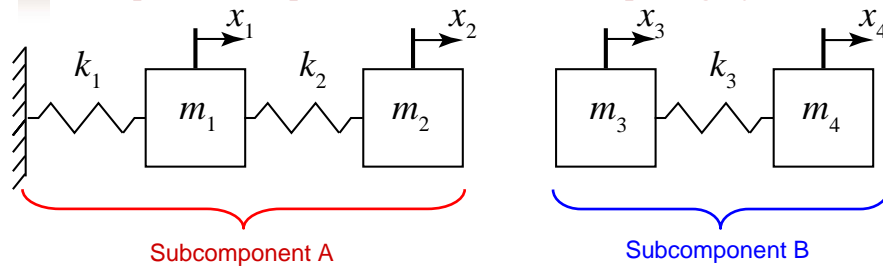
- Education research has consistently shown that we retain:



- Our goal in the next 30 minutes is to for you to begin to DO substructuring so the concepts will take root.



Simple Example: discrete mass-spring system



■ Exercise: $A+B=C$

- **Given:** Measurements of $H_{1,2}^A$, $H_{2,2}^A$, $H_{3,3}^B$, $H_{4,3}^B$ **Compute** $H_{1,4}^C$ using Frequency Based Substructuring (FBS)
 - Extra credit: Derive any expressions that you need by enforcing compatibility (equivalent displacements at the interface) and equilibrium (sum of forces at the interface).
- **Given:** Measured natural frequencies, ω_n damping ratios, ζ_n and mode shapes, ϕ_r for each subcomponent, **Compute** the natural frequencies and mode shapes of the assembly using Modal Substructuring (MS).

Discrete_Mass_Example.m

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Expanded Version of FBS Equation

■ Constraints:

$$\sum_{s=1}^{N_s} b^{(s)} u_c^{(s)} = 0$$

■ Response of assembly:

$$\begin{bmatrix} u_i^{(s)} \\ u_o^{(s)} \\ u_c^{(s)} \end{bmatrix} = Y^{(s)} f^{(s)} - Y^{(s)} \begin{bmatrix} 0 \\ 0 \\ b^{(s)T} \end{bmatrix} \left(\sum_{r=1}^{N_s} b^{(r)} Y_{cc}^{(r)} b^{(r)T} \right)^{-1} \sum_{r=1}^{N_s} [0 \quad 0 \quad b^{(r)}] Y^{(r)} f^{(r)}$$

From: D. J. Rixen, T. Godeby, and E. Pagnacco, "Dual Assembly of substructures and the FBS Method: Application to the Dynamic Testing of a Guitar," in *International Conference on Noise and Vibration Engineering: ISMA Lueven*, Belgium, 2006

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Additional Exercises:

- Frequency Based Substructuring
 - Add a small amount of random noise to your FRFs and repeat the computation. How sensitive is the solution to errors in your FRFs? Is it particularly sensitive at certain frequencies?
- Modal Substructuring
 - Check how your results vary if you exclude the elastic mode from your solution. Do the results make sense?
 - Add random errors to your identified natural frequencies, damping ratios and mode shapes. How sensitive is the substructuring solution to noise in the modal parameters?
- Once you are done, you can see my solution in:
"Discrete_Mass_Example_Solution.m"



Modal Substructuring Implementation

- Consider the EOM for two independent structures:

$$\begin{bmatrix} \mathbf{I}^A & 0 \\ 0 & \mathbf{I}^B \end{bmatrix} \begin{Bmatrix} \ddot{\mathbf{q}}^A \\ \ddot{\mathbf{q}}^B \end{Bmatrix} + \begin{bmatrix} -\omega_r^2 & 0 \\ 0 & -\omega_r^2 \end{bmatrix} \begin{Bmatrix} \mathbf{q}^A \\ \mathbf{q}^B \end{Bmatrix} = \begin{bmatrix} (\Phi^A)^T & 0 \\ 0 & (\Phi^B)^T \end{bmatrix} \begin{Bmatrix} \mathbf{f}^A \\ \mathbf{f}^B \end{Bmatrix}$$

$$\begin{Bmatrix} \mathbf{u}^A \\ \mathbf{u}^B \end{Bmatrix} = \begin{bmatrix} \Phi^A & 0 \\ 0 & \Phi^B \end{bmatrix} \begin{Bmatrix} \mathbf{q}^A \\ \mathbf{q}^B \end{Bmatrix}$$

- Recall that we connect them by enforcing some constraints:

$$\mathbf{B}_p \begin{Bmatrix} \mathbf{u}^A \\ \mathbf{u}^B \end{Bmatrix} = 0 \quad \mathbf{B} = \mathbf{B}_p \begin{bmatrix} \Phi^A & 0 \\ 0 & \Phi^B \end{bmatrix} \quad \mathbf{B} \begin{Bmatrix} \mathbf{q}^A \\ \mathbf{q}^B \end{Bmatrix} = 0$$

- Then eliminate some modal DOF using constraint equation to find \mathbf{L} and multiply matrices to compute the unconstrained EOM. The "ritzcomb.m" routine implements all of this.



Matlab Implementation of Modal Substructuring

- The “ritzscmb.m” Matlab® routine by M.S. Allen implements all of the calculations needed for modal substructuring.
- See ritzscmb_test_script_FE.m for an example of how to call the function.

```
% Structural modification using experimentally determined mode vectors as
% a Ritz vector basis set. Combines substructures (add or subtract)
% defined by modal parameters using the user supplied constraints and
% returns the modal parameters for the built-up system.
%
% [wn_mod,zt_mod,phi_mod,subsys_ind,coord_ind,M_hat,C_hat,K_hat,all_data] = ...
%   ritzscmb(subsys,cararray,sgn_vec);
%
% INPUTS:
%   subsys - N_sub dimensional array with the following fields for each
%   substructure: (i.e) subsys(1).wn = wn vector for subsystem #1.
%   wn - vector of natural frequencies for the un-modified structure.
%   zt - (optional) vector of modal damping ratios - set to zero
%   if not supplied.
%   phi - matrix of mass normalized mode vectors where each column
%   corresponds to a natural frequency in wn.
%   mmass - (optional) vector of modal masses - set to one if not
%   supplied. (Set these to zero for residual flexibilities.)
%   names - (optional) vector of node names for subsystem. These are
%   concatenated into a vector of names for phi_mod if supplied.
%
```



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Matlab Implementation (continued)

```
%   cararray - 3D array containing the constraints between subsystems. Each
%   constraint is a 3-column matrix with the following columns:
%   cararray(:,C_num) = [a1, pt1, subsys1;
%                       a2, pt2, subsys2;
%                       ...];
%   where a1 are the coefficients in an equation of the form:
%   a1*y1+a2*y2+...=0,
%   subsys1 is the index of the subsystem being joined (y1 is on
%   subsys1),
%   and pt1 is the index of the node in subsys1 that is being joined.
%   For Example: To assign node 3 on subsystem 2 to have the same
%   displacement as node 4 on subsystem 1, the following matrix would
%   be used: cararray = [1, 3, 2; -1, 4, 1]
%   NOTE: This assumes the coordinate systems are identical for
%   each component - use care if this is not the case!!
%   ##### SEE mkconstr for alternative #####
%
%   sgn_vec - [optional] vector of length N_sub of +1 or -1 values if each
%   substructure is being added or subtracted respectively to the
%   master system. All substructures are added if this is not
%   supplied.
%
% OUTPUTS:
%   Modal params of combined system: wn_mod, zt_mod, phi_mod
%   The rows in phi_mod correspond to a master set of coordinates
%   including each subsystem stacked sequentially. Thus, the coordinates
%   that have been joined will be repeated in this vector.
%   subsys_ind - vector telling which subsystem each node belongs to.
```



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Simplifying Input for “ritzscmb” → “mkconstr.m”

```
% Make constraints for ritzscmb - creates constraint arrays to join pairs
% of subsystems given the subsystem index and the names of the nodes to be
% joined.
%
%   carray = mkconstr(subsys,s1,s2,nodes,s1,s2,nodes,...)
%
% subsys is a cell array containing the following for the kth subsystem:
%   subsys(k).wn = vector of nat freqs
%   subsys(k).zt = vector of damping ratios (optional)
%   subsys(k).phi = mode shape matrix (No x N where No = number of outputs,
%                   N = number of modes)
%   subsys(k).names = cell array of names of each of the nodes (length No)
%
% The result is:
%   carray = [1, ns1, s1; -1, ns2, s2]; etc... as needed for ritzscmb.m
%
% EXAMPLE: Join two subsystems so that (y1)1=(y1)2 and (z1)1=(z1)2
displacement
%   subsys(1).phi = [3xN matrix]
%   subsys(1).names = {'x1','y1','z1'};
%   subsys(2).phi = [4xN matrix]
%   subsys(2).names = {'x3','x1','y1','z1'};
%   carray = mkconstr(subsys,1,2,{'y1','z1'})
%
%   returns the constraint array needed to enforce equal motion of
%   those two nodes between subsystems 1 and 2.
%   - One could then use [...] = ritzscmb(subsys,carray,[1,1]) to
%   compute the modes of the coupled system.
```



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Solution using “ritzscmb.m”

```
% Modal substructuring solution:
subsys(1).wn=fn_A*2*pi;
subsys(1).zt=zt_A;
subsys(1).phi=Phi_A;

subsys(2).wn=fn_B*2*pi;
subsys(2).zt=zt_B;
subsys(2).phi=Phi_B;

% Define array that enforces the constraint x2=x3 => 1*x2-1*x3=0;
carray=[1,2,1; -1,1,2]; % the interpretation is as follows:
% factor,coord_number,subsys_number
% where the factors are (1) and (-1) and we are connecting coord. 2 on
% subsystem 1 to coord 1 on subsystem 2.
[wn_MS,zt_MS,phi_MS] = ritzscmb(subsys,carray,[1,1])
% Compare those values with
fn_MS=wn_MS/2/pi

res=(phi_MS(4,:).*phi_MS(1,:)).'; % modal residues used to create FRFs for input
at 1 and output at 4

H41MS=cl_model(wn_MS,zt_MS,res,ws,'A');

figure(3)
semilogy(ws/2/pi,abs(H41C),ws/2/pi,abs(H41MS),'--'); grid on;
xlabel('\bfFrequency (Hz)'); ylabel('\bfFRF m/N');
title('\bfAssembly C: Modal Substructuring'); legend('H_{41}^C','H_{41}^C MS');
```



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Structural Modification: Example with ritzsmo.d.m

- Steel fixture (4.6"x1"x1" bar), treated as a mass and inertia added to the end of a 1"x0.75"x12" steel beam.

$$\text{ML} - \text{FF} = \text{FF}$$

- Simulated in Salinas using 40 beam elements.
 - Weight of Beam: 2.5 lb
 - Weight of Fixture: 1.3 lb
- "ritzsmo.d.m" used (via "ritzsmo.d_test_script_FE.m") used to remove mass and inertia.



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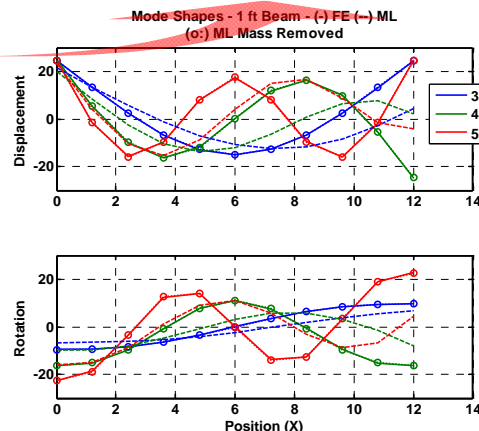
Structural Modification Example

Natural Frequencies (Hz) and Errors (%)						
Mode #	ML (FE)	FF (FE)	FF (Ritz Method)	E (%) Ritz	A FF (Sensitivity)	E (%) Sensitivity
3	703.766	1086.19	1085.32	-0.08%	835.54	-23.1%
4	1669.55	2988.13	2986	-0.07%	2055.7	-31.2%
5	3356.85	5845.21	5837.76	-0.13%	3673.9	-37.1%
6	6128.5	9639.3	17705.6	83.68%	6367	-33.9%

- Natural Frequencies and Mode Shapes from full (40 node) FE model compared with those estimated by "ritzsmo.d" using 6 modes (all modes below 6400 Hz).

- Errors in all but the 6th (last) mode are tiny.

Mode shapes are quite accurate
Sensitivity estimate of natural frequencies has large errors (30%) because the mass addition is not small.



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Joining Structures: Example Problem

- Ritz Method used to combine models for 12" and 24" beams using modes for each substructure up to 6400 Hz .
 - Case 1: Free-Free (FF) modes – modal parameters of each FF beam used to predict combined system response:

$$\text{A} \rightarrow y_c + y_c \text{B}$$

- Case 2: Mass-Loaded (ML) modes – parameters of modes with 4.6in end blocks used for each beam.

$$\text{---} \text{---} \text{---} + \text{---} \text{---} \text{---} - \text{---} \times 2$$



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Method

- Salinas models of 12" and 24" beams created with and without end blocks.
 - Each beam meshed with 0.3 inch long elements.
 - Solution of 12" and 24" beams joined was also found.
- The modal model for the case of Mass-Loaded modes was corrected using the "ritzsmode" script in order to remove the effects of the end masses. The modified modal models were then combined to form the 36-in beam.



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Correcting for End Mass and Inertia

Mode #	Natural Frequencies (Hz)							
	A ML	A ML-corr	B ML	B ML-corr	A FF	B FF	E (%) ML	E (%) ML
3	703.766	1085.32	213.845	271.945	1086.19	272.055	-0.08%	-0.04%
4	1669.55	2986	598.158	749.272	2988.13	749.554	-0.07%	-0.04%
5	3356.85	5837.76	1122.28	1468.17	5845.21	1468.62	-0.13%	-0.03%
6	6128.5	17705.6	1769.53	2425.59		2426.24		-0.03%
7			2646.01	3620.69		3622		-0.04%
8			3796	5053.16		5055.21		-0.04%
9			5205.86	13059.8				

- Chart compares Mass-Loaded natural frequencies with those obtained after analytically subtracting off the mass using "ritzmod.m" with the Free-Free natural frequencies for systems A and B.
- The natural frequencies predicted match the FF natural frequencies remarkably well.



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Natural Frequencies for 36-in Beam

Mode #	Natural Frequencies (Hz)				
	FEA	FF	ML	E (%) FF	E (%) ML
3	121.0	128.0	121.0	5.8%	0.00%
4	333.3	364.0	333.3	9.2%	-0.02%
5	653.3	658.8	653.3	0.8%	0.00%
6	1079.7	1130.8	1079.7	4.7%	0.00%
7	1612.4	1781.8	1612.4	10.5%	0.00%
8	2251.3	2282.0	2252.7	1.4%	0.06%
9	2996.3	3176.3	3003.1	6.0%	0.23%
10	3847.2	4396.0	3855.6	14.3%	0.22%
11	4803.7	4934.3	4836.4	2.7%	0.68%

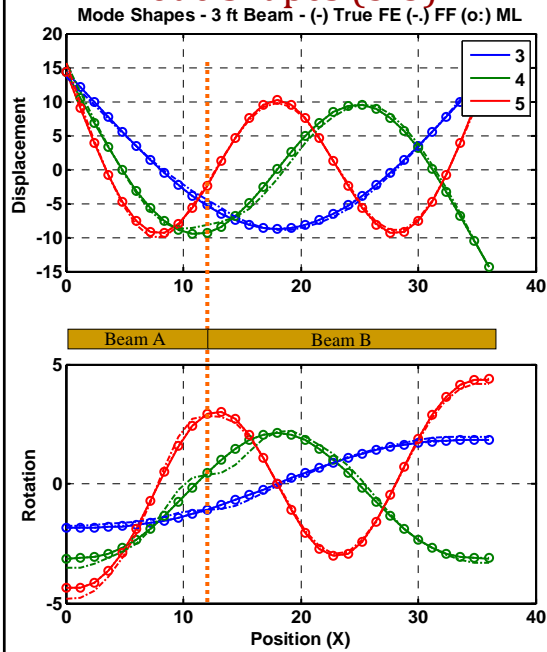
- Table shows natural frequencies of 36-in beam (1) predicted by FEA, (2) predicted using the Free-Free modes of the 12" and 24" beams and (3) predicted using Mass-Loaded modes of the 12" and 24" beams.
- Free-Free modes have errors ranging from 0% to 15%, while the Mass-Loaded modes show errors less than 1%.
- Error in natural frequency is largest for modes (of 36-in beam) that have large shear (derivative of rotation) at the connection point. (see mode shapes on next slide)



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Mode Shapes (3-5)

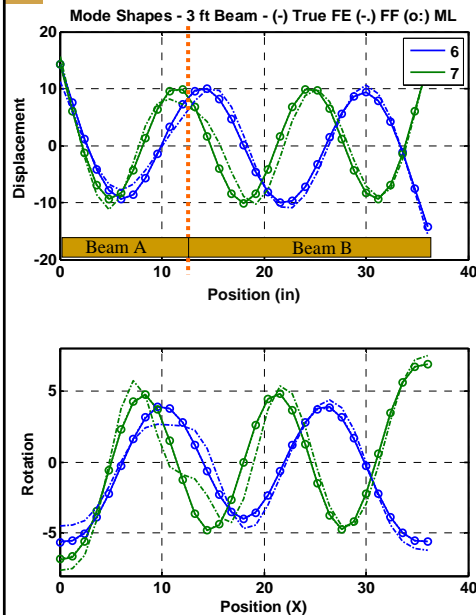


- Mass loaded mode vectors match the analytical mode vectors more closely, especially the rotation component at the connection.
- Note: shear and moment in beam proportional to d^2u/dx^2 and d^3u/dx^3 .
- Free-Free modes all have zero shear and moment forces at the free end, so they cannot represent the combined structure well near the connection.

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Mode Shapes (6-7)



- Shapes of modes 6 and 7 for the 36" beam.
- The FF modes are clearly not a very good basis for these modes.

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Component Mode Synthesis

- There are a few modal bases that are commonly used in analytical substructuring.
 - Craig-Bampton Method:
 - Fixed interface modes + Constraint Modes
 - Efficient basis substructures that are connected rigidly
 - Rule of thumb: Include modes out to 1.5 to 2.0 times the frequency range of interest.
 - Method of MacNeal (variations by Rubin, etc...)
 - Free-interface modes + (variations on) Residual Flexibility
 - Has been found to give good results when subcomponents are weakly coupled (e.g. mounted on vibration isolation mounts).
 - Mass-loaded interface modes
 - Provide a compromise between free-free and fixed-interface modes



[1] R. R. J. Craig, *Structural Dynamics*. New York: John Wiley and Sons, 1981

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